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Performance Reports:

AXAF SIM Focus Mechanism Study

Prepared in accordance with DRD# 784MA-002

Principal Investigator

Dr. H. D. Tananbaum

Prepared for:
George C. Marshall Space Flight Center
ational Aeronautics and Space Administration
Marshall Space Flight Center, AL 35812

Submitted by:
Smithsonian Astrophysical Observatory
60 Garden Street
Cambridge, MA 02138

The Smithsonian Astrophysical Observatory
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ORIGINAL CONTAINS
COLOR ILLUSTRATIONS

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Date: 16 Feb 1994

Document No.: SAO-AXAF-DR-94-081

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Revision Record

Revision	Date	DCO No.	Affected Pages
"A"	16 Feb 94	-------	Initial Release

AXAF SIM Focus Mechanism Study

E. Whitbeck

Smithsonian Astrophysical Observatory
60 Garden St.
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16 February 1994

Summary

Analysis of the SIM focus mechanism (small motion version) has shown the following:

1. The focus motion vs. focus ring rotation relationship is fairly linear and can be easily calibrated.

2. The flexure stresses are high and will be higher yet in the large motion version. These should be carefully examined in the design process.

3. The SIM alignment (tilt) is insensitive to normal manufacturing and assembly tolerances.

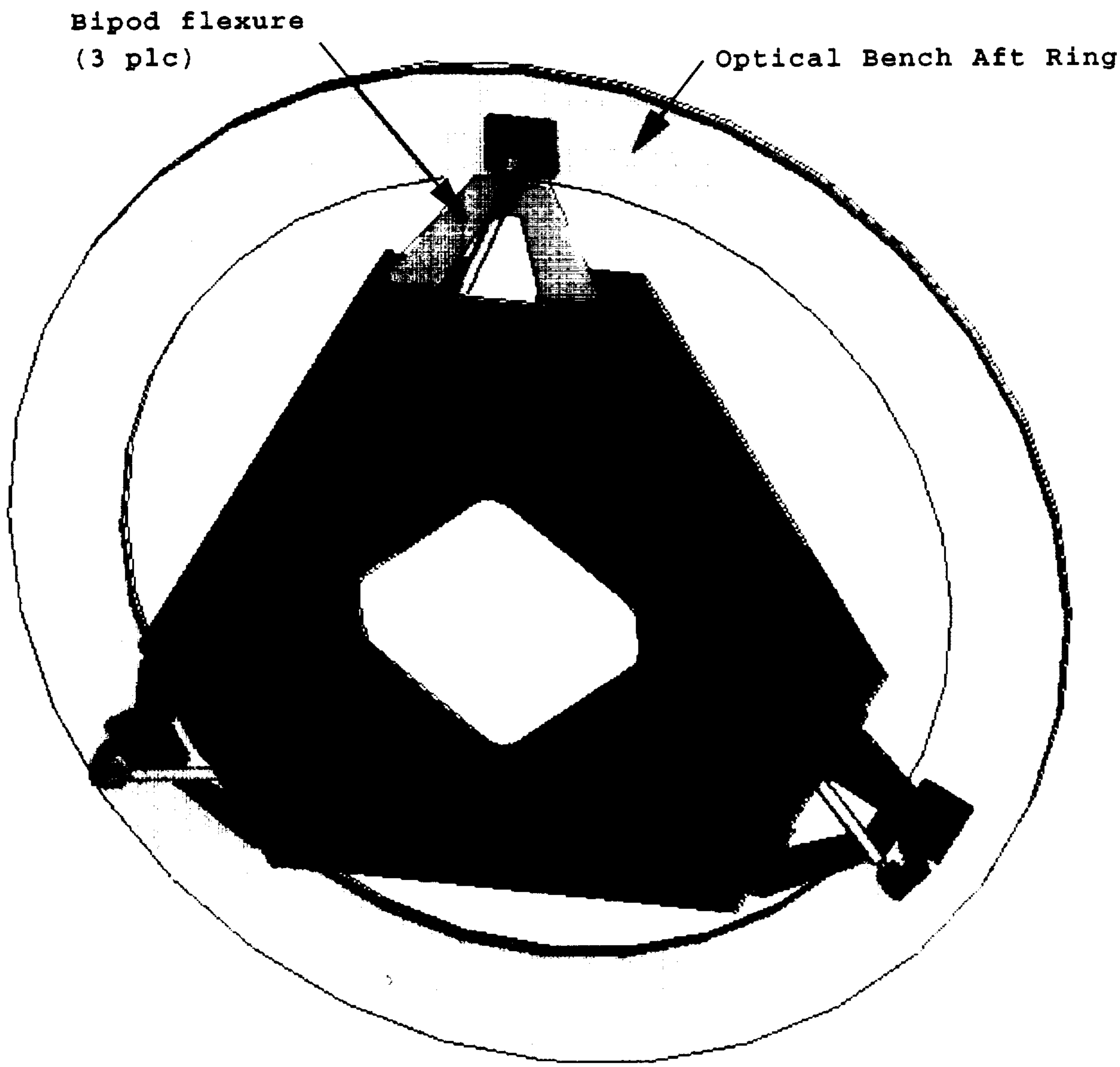
4. The tilt resulting from a c.g. shift due to changing instruments in the focal plane is < 1 arcmin and should present no problem.

1. Introduction

The design requirements and initial design concept for the AXAF-I Science Instrument Module(SIM) were reviewed at Ball on September 29, 1993. The concept design SIM focus mechanism, shown in Figures 1,2 and 3, utilizes a planetary gearset, with redundant motors, to drive a large ring(called "Main Housing Bearing") via a spur gearset. This large drive ring actuates three tangent bar links(called "Push Rods"), which in turn actuate three levers(called "Pin Levers"). Each of the three Pin Levers rotates an "Eccentric Pin", which in turn moves the base of a bipod flexure in both the radial(normal to optical axis and axial (focus along optical axis) directions. Three bipod flexures are employed, equally spaced at 120 degrees apart, the base of each being translated in the two directions as described above. A focus adjustment is made by rotating the drive ring, which drives the push rods and therefore the pin levers, which in turn rotate the eccentric pins, finally imparting the two motions to the base of each of the bipod flexures. The axial translation (focus adjustment) of the focussed structure is the sum of the direct axial motion plus axial motion which comes from uniformly squeezing the three bipod bases radially inward. SAO documented the following concerns regarding the focus mechanism in memo WAP-FY94-001, dated October 7, 1993:

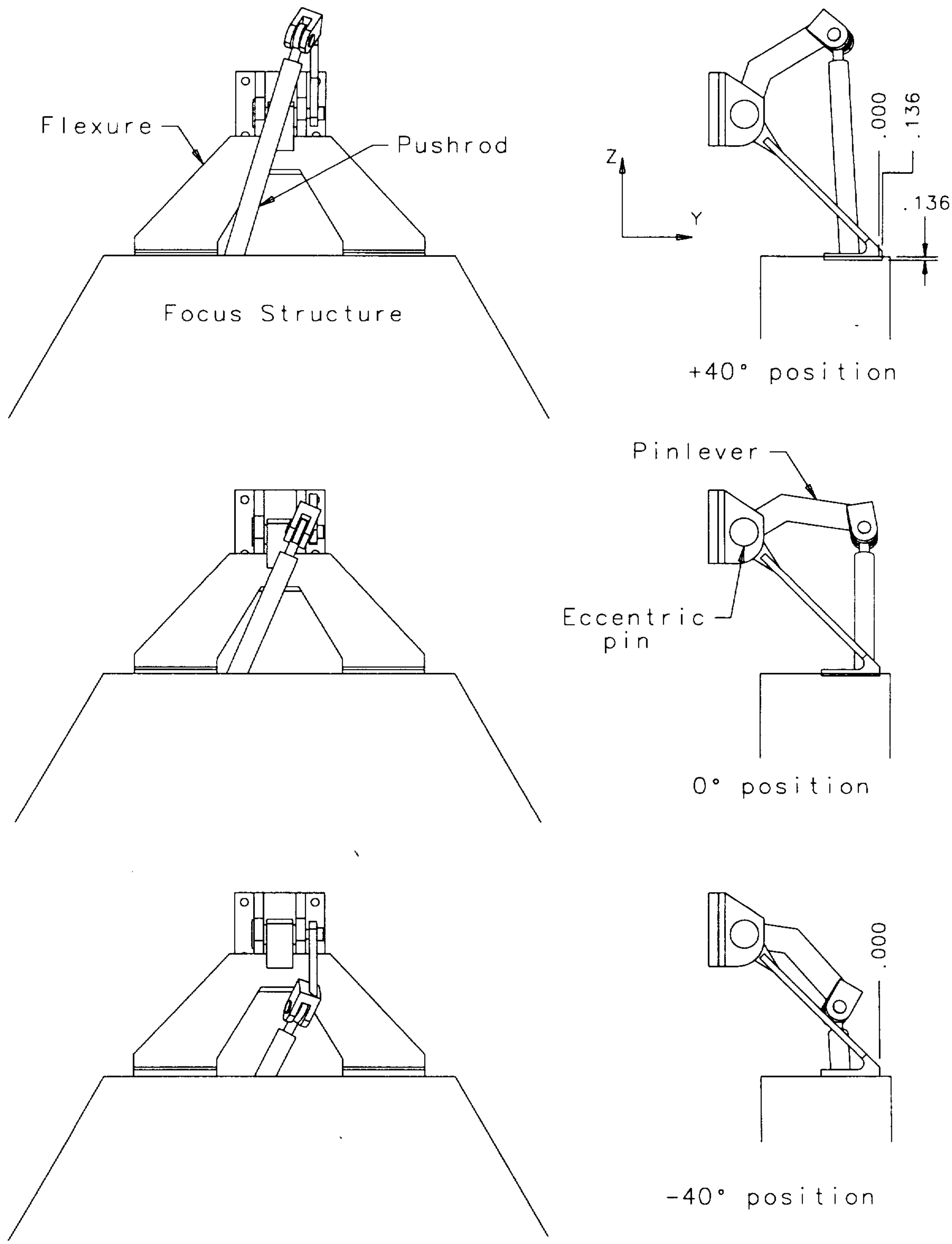
1) The focus adjustment depends, in large part, on the structural properties (stiffnesses and end fixities) of the bipod flexures, push rods, pin levers and eccentric pins. If these properties are not matched very well, then lateral translations as well as unwanted rotations of the focussed structure will accompany focus motion. In addition, the stackup of linkage tolerances and any non-uniform wear in the linkages will result in the same unwanted motions. Thermal gradients will also affect these motions. At the review Ball did not present supporting analyses to support their choice of this design concept.

2) The proposed "primary" method of measuring focus is by counting motor steps. The "backup" method is by a pot mounted on the drive ring. Neither method provides for a direct measurement of the quantity desired(focus position). This is of concern because of the long and indirect relationship between focus and the sensed quantity(drive ring rotation). There are three sinusoidal relationships and structural stiffness in the path, and the resulting calibration is likely to be highly nonlinear. These



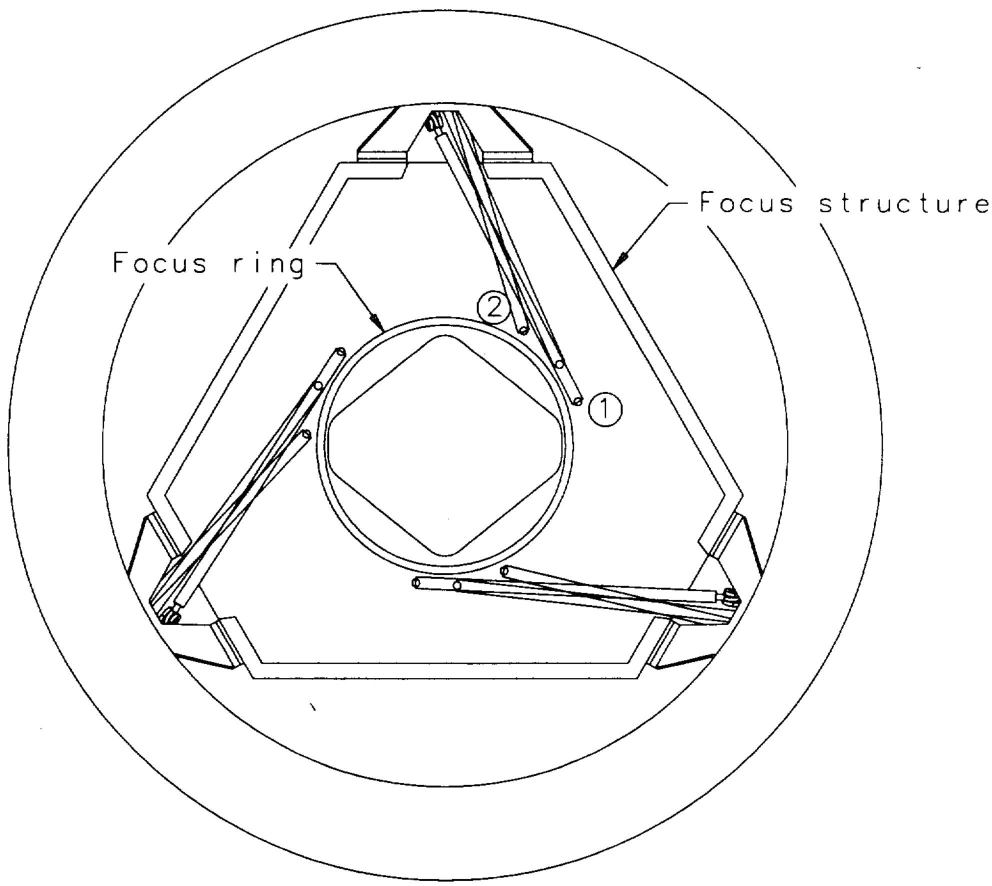
SIM Focus Mechanism

Figure 1



Rigid body motion of flexure
(flexure unattached to focus structure)

Figure 2



Rotation of focus ring
showing extreme and middle
positions of pushrods

Figure 3

methods would require an accurate ground calibration

3) Ground calibration(and verification) of focus vs.drive position must be done in 1G on the ground. This calibration will be complicated by both the structural characteristics of the bipods and the fact that the CG of the translating portion of the SIM is not on the optical axis(thereby causing unwanted rotations and changing the focus position vs. motor step and pot readout relationships). The SIM translating weight could be offloaded, but the calibration then becomes sensitive to any errors in offloading(both magnitude and direction). There are concerns as to whether a calibration to the required accuracy(better than 0.0005") can be accomplished on the ground.

4) The choice of a potentiometer as the focus position sensor is questionable in terms of reliability for a five year mission life. In addition, Rob Cameron's recent negative experience with this same sensor on the GRO OSSE experiment raises our level of concern.

The results of SAO's study of items 1, 2 and 3 described above are presented in this report.

2. Methodology

Evaluation of the focus mechanism was performed as follows:

1. An Ideas solid model of the mechanism was created to visualize the geometry of the mechanism and its motion.

2. A semi-analytical relationship between focus motion and drive ring rotation was developed. This was used to address concerns over the indirect drive and its functional form.

3. A single flexure Ideas finite element model was created to evaluate flexure axial motion vs. radial inward "squeeze" of the flexure base.

4. A three flexure Ideas finite element model was developed and run to evaluate the performance of the overall focus drive mechanism with respect to structural non-uniformities and calibration in 1G.

3. Results

Solid Model

As a first step an Ideas solid model of the AXAF SIM focus mechanism was made from information supplied by Ball in the SIM Concept Audit, 9/29/93. Dimensions which were not explicitly given were scaled or estimated from the pictures. The pin lever was angled to prevent it from bumping into the flexure in the -40 degree position, otherwise the mechanism is as close to Ball's as could be determined. (See fig. 1). The range of motion was taken to be -40 degrees to +40 degrees rotation of the pin lever and eccentric pin, with the neutral position being the pin lever parallel to the line of focus and the top of the flexure -45 degrees from the parallel to the line of focus through the eccentric pin. An animated sequence which shows the working of the mechanism was also developed. The extreme and neutral positions are shown in figures 2 and 3.

Deformation of Flexure

Deformation of the flexure yields a significant component of the focus motion. The eccentric pin in placed so that the flexure is preloaded and increasingly loaded over the full range of motion. To first order the motion of the flexure is that of a rigid body with the same geometry, i.e. since the flexure is at a 45 degree angle a motion of one inch in the radial direction at the pin end will produce a motion of one inch in the axial direction at the foot end. The motion is not a function of flexure EI.

Inspection of the mechanism has shown that the radial motion of the flexure at the pin is .136". The resulting deformation of the flexure was verified by finite element analysis. A FEM of the flexure was generated from an Ideas solid model (see fig. 4). The eccentric pin is modeled as a node at the pin center with rigid beams to the pin surface, which in turn are connected to the flexure with circumferentially soft and radially hard springs. The bottom of the flexure foot is free to slide in the focus direction and is otherwise fixed. A .136" radial motion is input, the expected .136" axial motion results (see fig. 5). It should be noted that the analysis performed was a standard linear small deflection analysis. These motions could properly be classified as large deflections. Since focus will not be directly measured and will be predicted in part by this structural analysis, a nonlinear large deflection analysis should be performed as part of the design process.

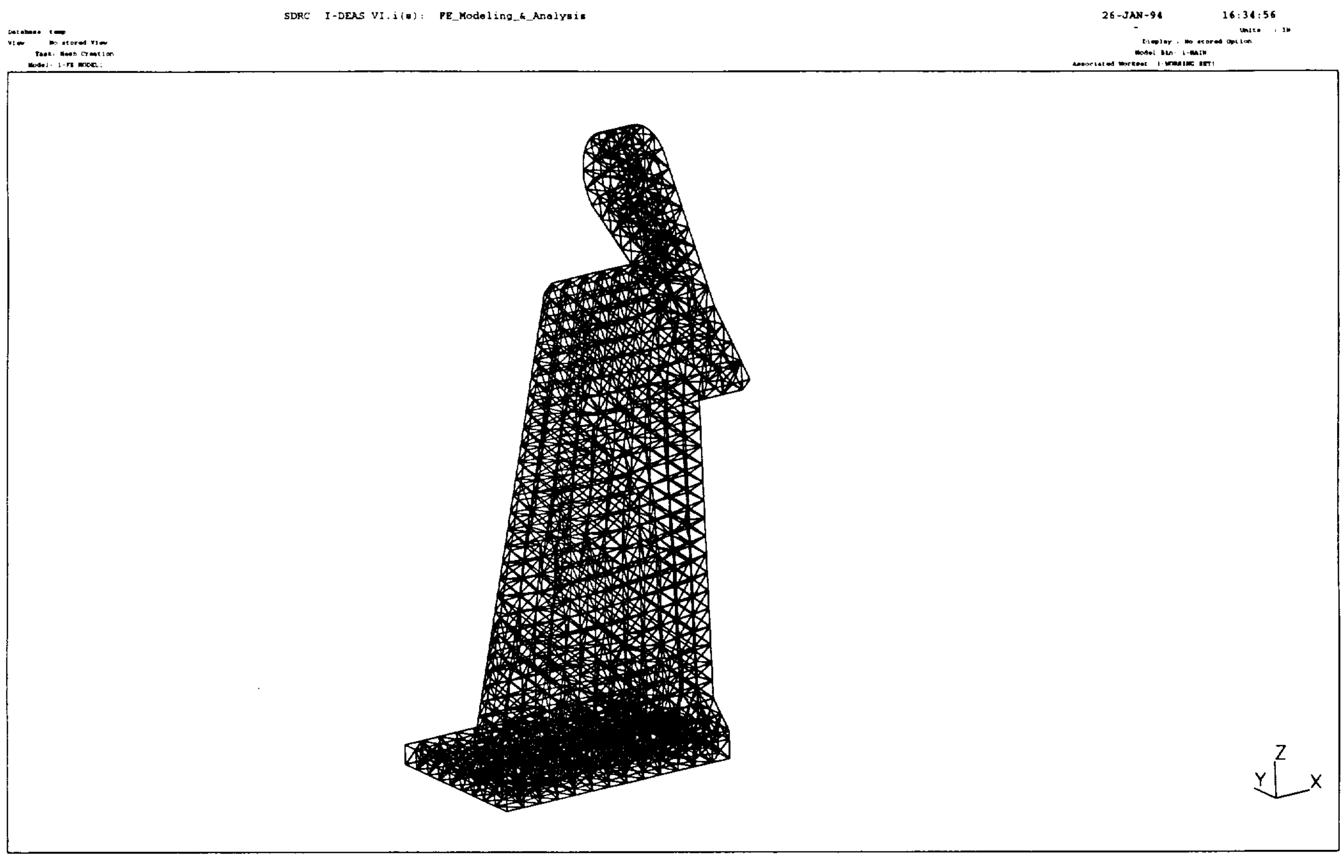


Figure 4 Flexure Finite Element Model

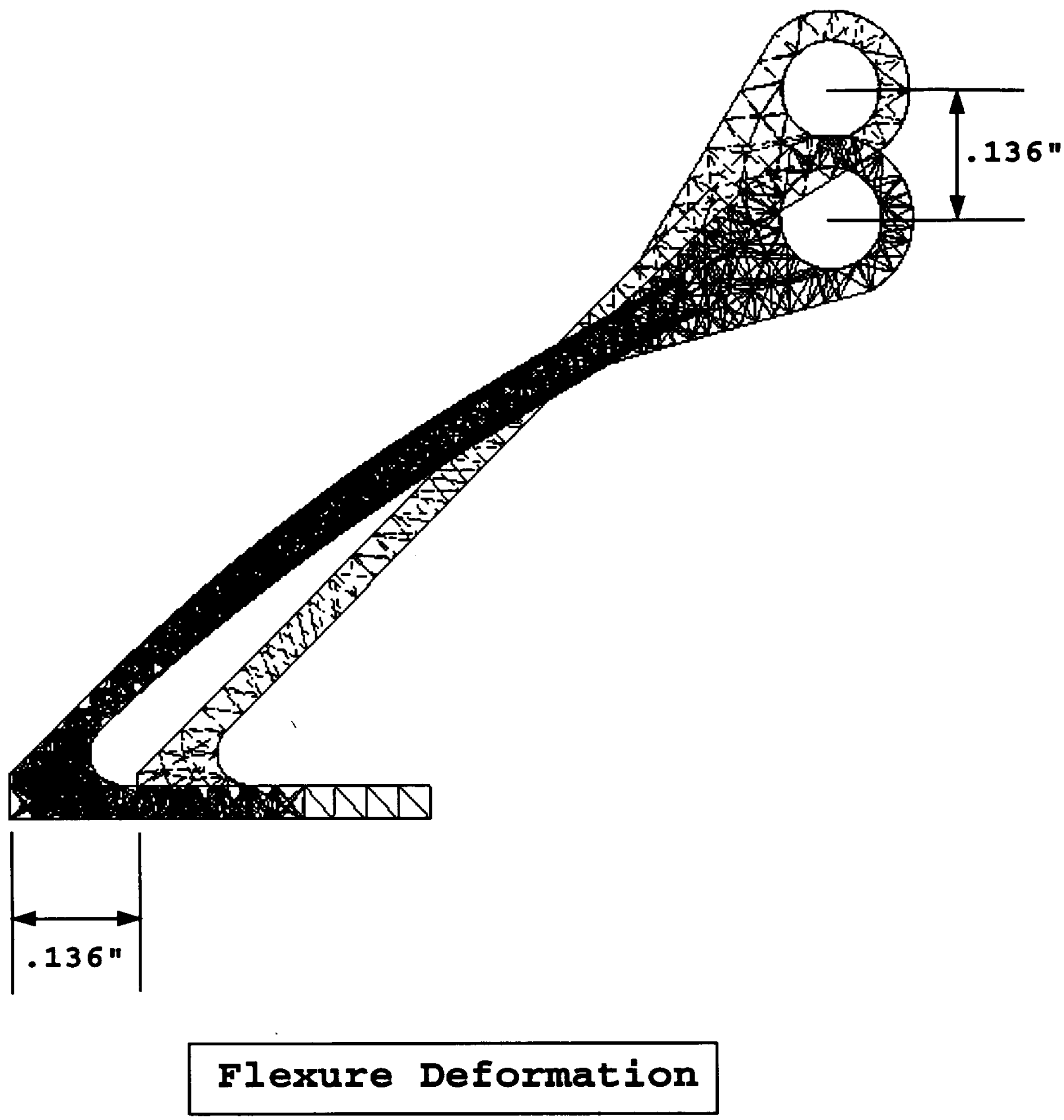


Figure 5

Of interest is the stress generated by the input load. The maximum stress in the flexure is about 24000 psi (see fig. 6). Since the requirement for the range of motion has increased by a factor of three over that used in this study, stress could become a design driver and should be carefully considered.

Motion due to flexure deformation is combined with flexure rigid body motion to determine overall focus motion.

Calculation of focus motion

Dimensions of the focus mechanism were scaled or estimated from the information supplied by Ball in the SIM Concept Audit, 9/29/93. It was assumed that the flexure would be prestressed so that at no point within the full range of focus motion would the flexure pass through an unstressed condition. (This is very important so as to eliminate potential backlash.)

A solid model of the mechanism was made to visually demonstrate the way the mechanism worked but it was necessary to calculate the motions in order to get an accurate graph of focus motion versus rotation of the focus ring. Focus is related to rotation of the focus ring by a fourth order equation. I found an approximate solution (accurate to .001 degree) by relating both focus and rotation of the focus ring to rotation of the eccentric pin at one degree intervals of focus ring rotation. (See figures 7 - 10 for details.) The relationship between focus motion and ring rotation is nearly linear and should present no significant problem of the type envisioned.

Sensitivity Analyses

A series of analyses were performed to determine the sensitivity of focus location and tilt due to mislocation of the flexures during manufacture, extremes of manufacturing tolerance, or to shifts in the center of gravity of the SIM. A finite element model comprising all three flexures connected by rigid arms to a central node was made (fig. 11).

Modifications to this model were made by moving one of the flexures a certain amount, then an analysis was made by applying the -.136" radial displacement to all three flexures. The resulting motion of the center of the focal plane was predicted.

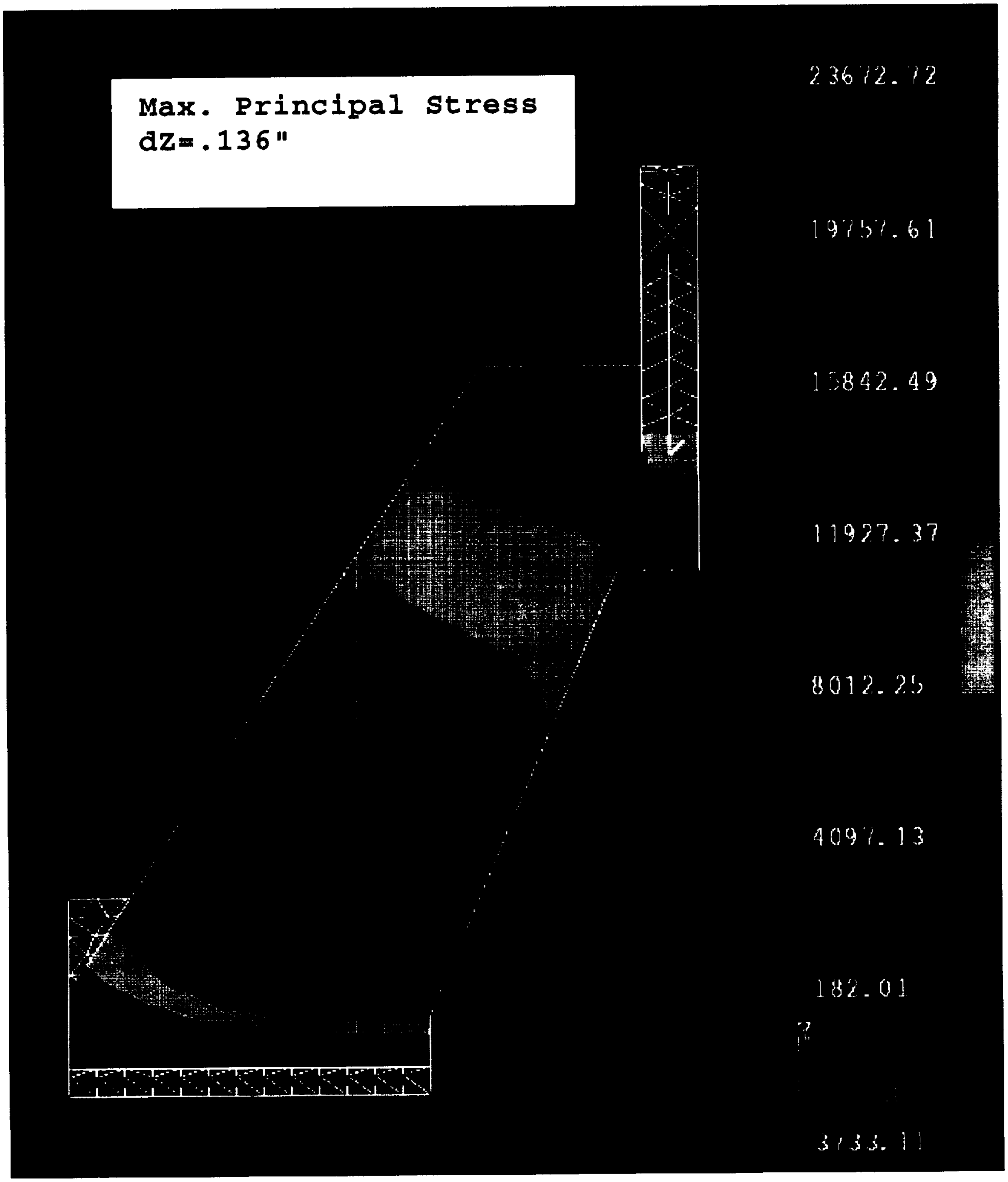


Figure 6 Flexure Stress Contours

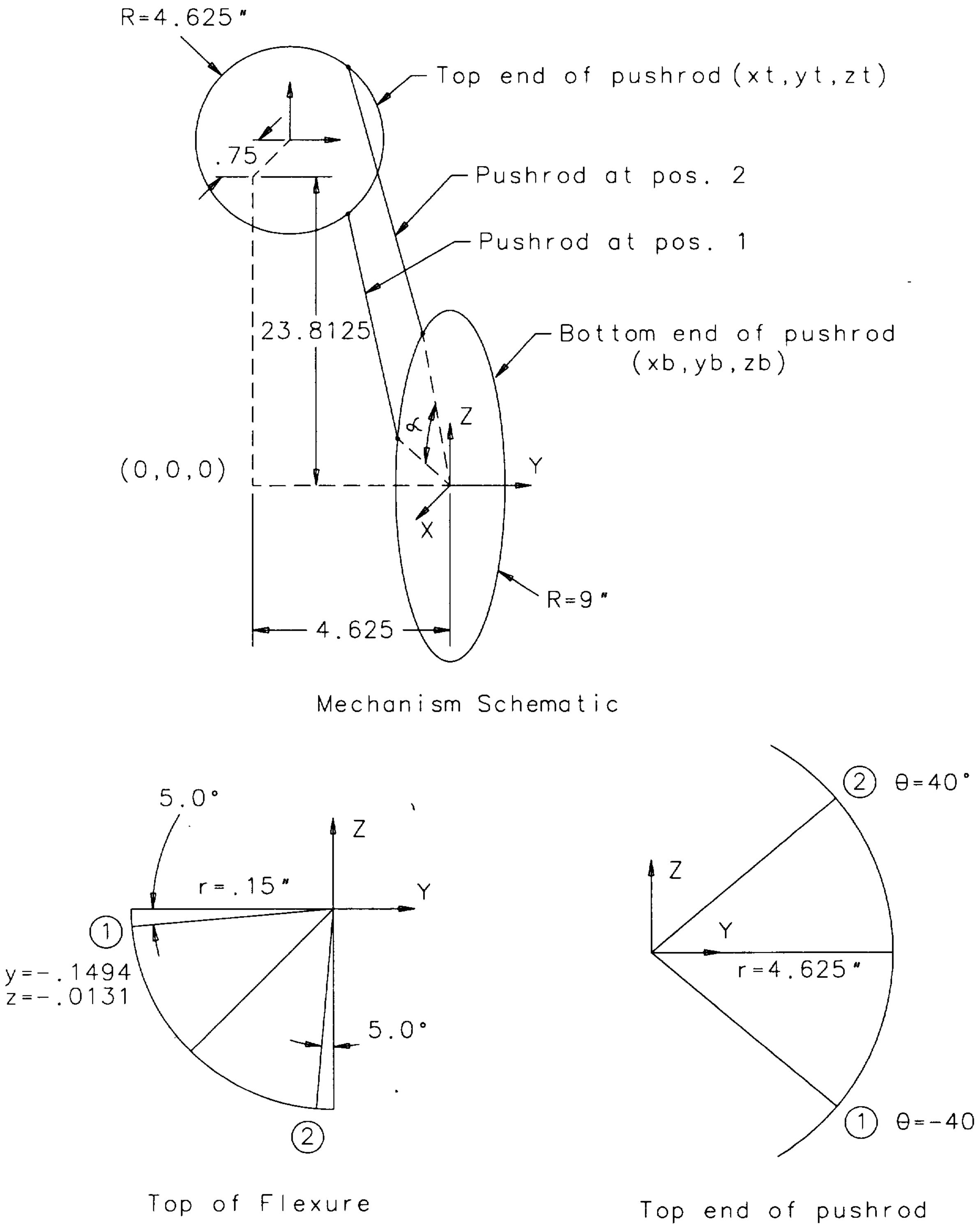


Figure 7 Mechanism Geometry

$l_{pushrod} = 20.34''$

$x_t = -.75''$

$y_t = 4.625\cos\theta$

$z_t = 23.8125 + 4.625\sin\theta$

rigid body focus motion = $.1494 - .15\cos(\theta+45)$

flex induced focus motion = $-.0131 + .15\sin(\theta+45)$

$$y_b = 4.625 + .1494 - .15\cos(\theta+45) - .0131 + .15\sin(\theta+45)$$

$$x_b = (81 - z_b^2)^{1/2}$$

$$(x_t - x_b)^2 + (y_t - y_b)^2 + (z_t - z_b)^2 = 20.34^2$$

Solve for z_b at different values of θ

$$C_1 = 20.34^2 - (y_t - y_b)^2 - x_t^2 - z_t^2 - 81$$

$$z_b^2(4z_t^2 + 4x_t^2) + z_b(4C_1 z_t) + (C_1^2 - 4x_t^2(81)) = 0$$

Solve for α (rotation of focus ring)

$$z_b = 9\sin\alpha$$

Figure 8 Calculation of Focus Motion

theta	thetarad	yt	zt	yb	C1	a	b
-40.107	-0.7	3.537399	20.833	4.624341	-103.042	1738.305	-8586.71
-37.215	-0.64952	3.68322	21.01527	4.632701	-110.39	1768.816	-9279.49
-34.453	-0.60132	3.813733	21.196	4.640991	-117.802	1799.332	-9987.7
-31.795	-0.55493	3.930968	21.37567	4.649231	-125.282	1829.928	-10712
-29.227	-0.51011	4.036202	21.55425	4.657422	-132.818	1860.593	-11451.2
-26.731	-0.46654	4.130719	21.73217	4.665582	-140.42	1891.398	-12206.5
-24.3	-0.42411	4.215241	21.90925	4.673705	-148.072	1922.311	-12976.6
-21.925	-0.38266	4.29049	22.08556	4.681791	-155.772	1953.338	-13761.2
-19.596	-0.34201	4.357124	22.26134	4.689854	-163.525	1984.519	-14561.1
-17.31	-0.30212	4.415529	22.43637	4.697882	-171.317	2015.813	-15375
-15.06	-0.26285	4.466151	22.61079	4.705882	-179.152	2047.24	-16203.1
-12.843	-0.22415	4.509296	22.78445	4.713847	-187.02	2078.775	-17044.6
-10.653	-0.18593	4.545287	22.95752	4.721785	-194.926	2110.441	-17900.1
-8.49	-0.14818	4.574318	23.12968	4.729681	-202.853	2142.178	-18767.7
-6.348	-0.11079	4.596643	23.30113	4.737545	-210.809	2174.02	-19648.4
-4.226	-0.07376	4.612425	23.47168	4.745368	-218.784	2205.929	-20540.9
-2.124	-0.03707	4.621822	23.64109	4.753138	-226.765	2237.854	-21443.9
-0.037	-0.00065	4.624999	23.80951	4.760863	-234.758	2269.822	-22357.9
2.035	0.035517	4.622083	23.97673	4.768533	-242.752	2301.785	-23281.6
4.095	0.071471	4.613192	24.14277	4.776148	-250.747	2333.744	-24214.9
6.144	0.107233	4.598434	24.3075	4.783704	-258.736	2365.669	-25156.9
8.181	0.142785	4.577934	24.47064	4.791186	-266.705	2397.499	-26105.7
10.21	0.178198	4.551762	24.63231	4.798602	-274.659	2429.253	-27061.9
12.23	0.213454	4.520036	24.79224	4.805937	-282.584	2460.871	-28023.6
14.242	0.24857	4.482852	24.95033	4.813188	-290.475	2492.326	-28989.8
16.25	0.283616	4.440231	25.10671	4.820361	-298.338	2523.637	-29961.2
18.25	0.318522	4.392359	25.26088	4.827432	-306.148	2554.699	-30934.3
20.245	0.353342	4.339275	25.41291	4.834405	-313.908	2585.514	-31909.3
22.235	0.388074	4.281084	25.56263	4.841272	-321.609	2616.042	-32884.6
24.222	0.422753	4.217828	25.71001	4.848032	-329.249	2646.269	-33860
26.205	0.457363	4.149643	25.85483	4.854674	-336.816	2676.138	-34833.3
28.186	0.491938	4.076563	25.99705	4.861197	-344.309	2705.636	-35804.1
30.163	0.526443	3.998774	26.13638	4.867588	-351.712	2734.692	-36769.9
32.138	0.560913	3.916309	26.27281	4.873846	-359.025	2763.293	-37730.3
34.112	0.595366	3.829237	26.40626	4.879966	-366.241	2791.411	-38684.2
36.084	0.629784	3.737715	26.53649	4.885939	-373.35	2818.991	-39629.6
38.055	0.664184	3.641816	26.66343	4.891762	-380.348	2846.004	-40565.5
40.025	0.698567	3.54166	26.78694	4.897427	-387.225	2872.41	-41490.3

Figure 9 Calculation of Focus Motion Spreadsheet

c	det	zb1	zb2	xb	alpha	alphadeg	
10435.44	1082.435	2.781198	2.158503	8.559494	0.314165	18.000	
12003.67	1086.183	2.930116	2.316043	8.509666	0.331613	19.000	
13694.99	1089.412	3.078118	2.472665	8.457257	0.349058	20.000	
15513.39	1092.134	3.225292	2.628474	8.402231	0.366517	21.000	
17458.49	1094.352	3.371388	2.783214	8.344684	0.383964	22.000	
19535.53	1096.07	3.516603	2.937101	8.284534	0.401429	23.000	
21743.13	1097.286	3.66067	3.089854	8.221891	0.418884	24.000	
24082.67	1097.998	3.803552	3.241439	8.156776	0.436331	25.000	
26558.16	1098.204	3.945374	3.391989	8.08913	0.45379	26.000	
29167.4	1097.898	4.08591	3.541267	8.019061	0.471238	27.000	
31913.18	1097.078	4.225235	3.689354	7.946533	0.488691	28.000	
34794.27	1095.737	4.363228	3.836121	7.87161	0.506138	29.000	
37813.82	1093.868	4.499989	3.981676	7.794235	0.523597	30.000	
40967.15	1091.468	4.635277	4.125764	7.714545	0.541044	31.000	
44258.32	1088.529	4.769254	4.268555	7.632445	0.558503	32.000	
47684.35	1085.045	4.901787	4.409911	7.548012	0.575963	33.000	
51240.18	1081.015	5.032704	4.549645	7.46136	0.593408	34.000	
54929.2	1076.428	5.162157	4.687922	7.37239	0.610861	35.000	
58746.33	1071.281	5.290001	4.824588	7.281201	0.628309	36.000	
62691.78	1065.567	5.416291	4.9597	7.187753	0.645766	37.000	
66762.01	1059.28	5.540965	5.093193	7.092088	0.663227	38.000	
70949.1	1052.423	5.663851	5.224884	6.99434	0.680674	39.000	
75255.07	1044.983	5.785088	5.354921	6.894401	0.698132	40.000	
79671.45	1036.963	5.904519	5.483139	6.792397	0.715583	41.000	
84193.53	1028.36	6.022117	5.609506	6.688356	0.73303	42.000	
88823.43	1019.157	6.138029	5.734185	6.582143	0.750498	43.000	
93544.54	1009.373	6.251947	5.856842	6.474037	0.767948	44.000	
98356.06	998.9971	6.363971	5.977588	6.363951	0.7854	45.000	
103249.9	988.031	6.474033	6.096351	6.251952	0.802847	46.000	
108222.5	976.4667	6.582178	6.213181	6.137991	0.820304	47.000	
113262.7	964.3131	6.688292	6.327954	6.022188	0.837756	48.000	
118366.5	951.5618	6.792422	6.440726	5.90449	0.855217	49.000	
123519.3	938.2307	6.89441	6.551326	5.785076	0.872666	50.000	
128716.4	924.3122	6.994309	6.659813	5.663889	0.890117	51.000	
133950.4	909.8049	7.09212	6.76619	5.540924	0.907575	52.000	
139208.3	894.7228	7.187744	6.870353	5.416303	0.925029	53.000	
144482.1	879.0661	7.281186	6.972308	5.290022	0.942484	54.000	
149760.9	862.8431	7.372403	7.072013	5.162139	0.959938	55.000	

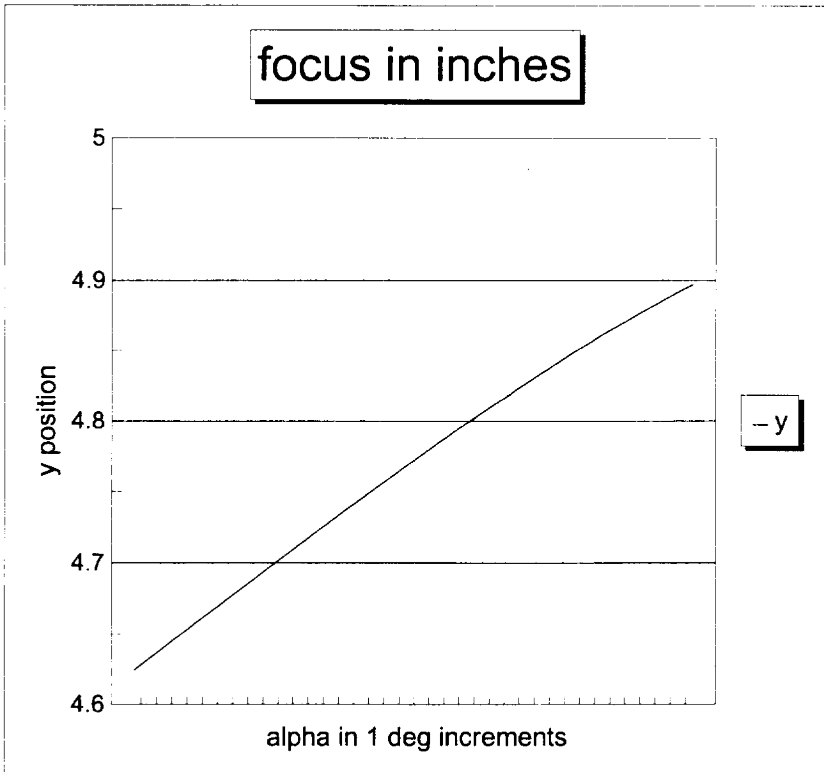


Figure 10 Focus Motion vs. Rotation of Focus Ring

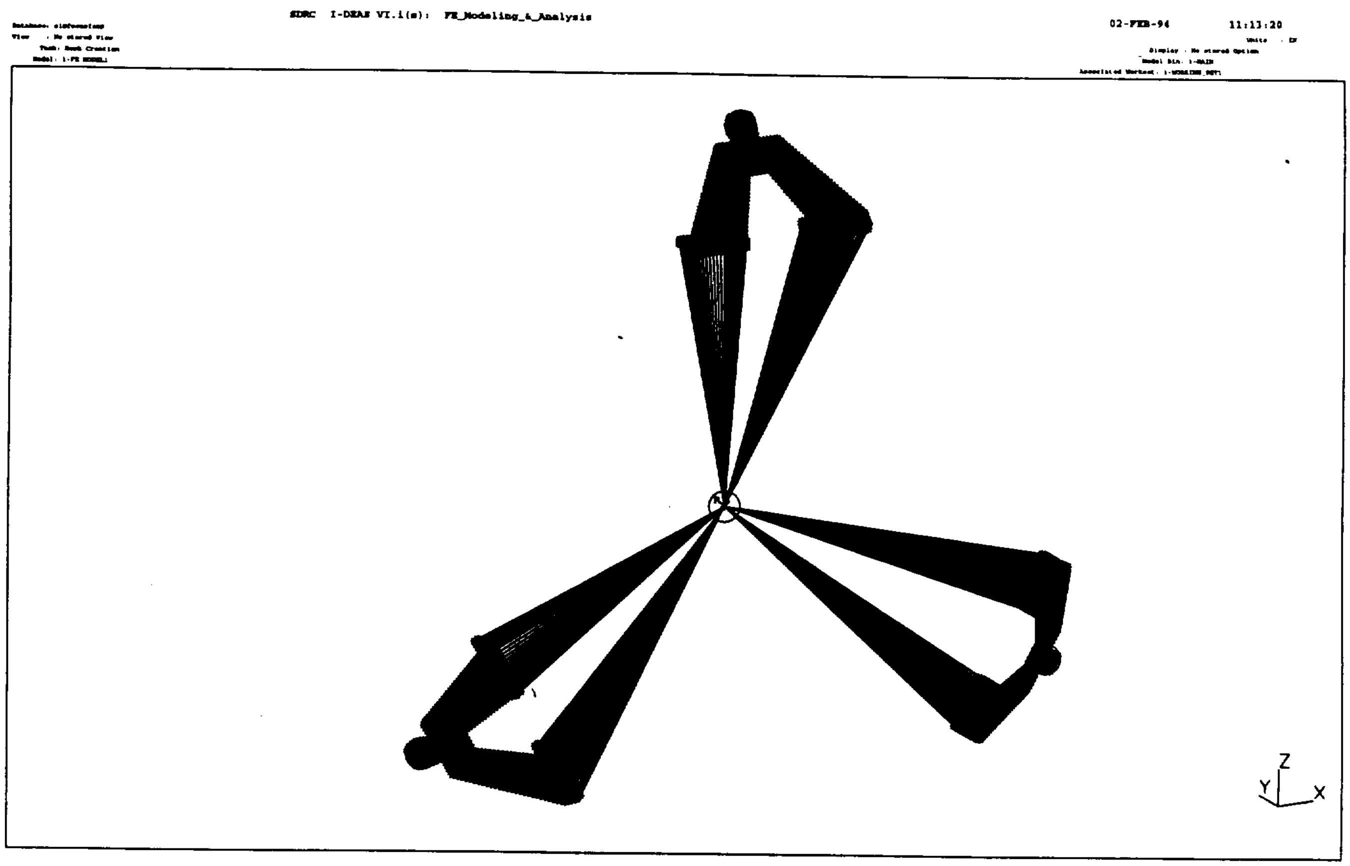


Figure 11 Three Flexure Finite Element Model

Flexure motion	Focal plane tilt
.01" +x	<.1 arcsec
.1 degree rotz	.6 arcsec
.01" -z	<.1 arcsec

Tilts of the focal plane on order of 10 arcsec are of concern. Combinations of flexure motion would yield larger tilts than those listed here but still not of 10 arcsec order. The results indicate that placement of the flexures to careful manufacturing tolerances will be satisfactory.

An analysis was made to determine the effect of a change in thickness of .01" of one of the flexures.

Condition	Focal plane tilt
One flexure 0.01" thinner	<.1 arcsec

An analysis was made of the effect on gravity sag of shifting the SIM c.g. by .5" during alignment in the vertical configuration (gravity acting along the mirror axis). Weight of the SIM is assumed to be 888 lbs, c.g. at 2" -y, 4.5" -z (Ball coordinate system).

Condition	Focal plane tilt
Nominal	4.4 arcsec
c.g. moved	4.9 arcsec

Again there is no significant effect provided the c.g. is known to within a fraction of an inch. The nominal case was calculated for the c.g. at one extreme of an approximately 11 inch travel. The overall tilt will be less than 1 arcmin for any configuration.

Model Accuracy

Several different flexure finite element models were made in order to determine the appropriate mesh density for this analysis. The initial model was a half flexure model of parabolic tetrahedra with about 7200 nodes. Another model was made using the same number of elements but linear instead of parabolic. This model had 1200 nodes. A third model was made of parabolic tetrahedra with two elements through the thickness of the flexure blade. This one had about 25000 nodes. All three models predict deflections within .2% of each other. The coarsest model was inappropriate for stress. Stresses between the middle and dense models varied by about 4%. The initial model (middle density) was deemed to be adequate for our stress calculation. The least dense model was used as a basis for the three flexure model where no stress calculations were performed.

4. Conclusions

Analysis of the SIM focus mechanism (small motion version) has shown the following:

1. The focus motion vs. focus ring rotation relationship is fairly linear and can be easily calibrated.

2. The flexure stresses are high and will be higher yet in the large motion version. These should be carefully examined in the design process.

3. The SIM alignment (tilt) is insensitive to normal manufacturing and assembly tolerances.

4. The tilt resulting from a c.g. shift due to changing instruments in the focal plane is < 1 arcmin and should present no problem.